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SUMMARY

A hydrogen fuel-flow valve with an electrohydraulic servosystem is described. An analysis of the servosystem is presented, along with a discussion of the limitations imposed on system performance by nonlinearities. The response of the valve to swept-frequency inputs is experimentally determined and compared with analytical results obtained from a computer model. The valve is found to perform favorably for frequencies up to 200 Hz.

INTRODUCTION

The hydrogen fuel-flow valve described herein is for use in the Modified Government Baseline (MGB) engine model test, which is part of the National Aerospace Plane (NASP) project. NASP is a joint NASA and Department of Defense project that proposes the development of a hypersonic aircraft that can take off from a runway to achieve space orbit. Because of its complexity, NASP requires a significant amount of research and technological development—as does the MGB engine model. A high-performance fuel-flow valve is needed for the control of this complicated dynamic system. Such a valve was designed and fabricated to meet NASA-defined specifications and the project's stringent requirements. Verification testing of the valve's dynamic performance was performed prior to actual use within the MGB engine model. A test bed was set up consisting of the fuel-flow valve, the electrohydraulic servosystem, and data-recording devices. This paper will describe the valve, servosystem, dynamics, and test-bed setup; present an analysis of the servosystem, including a discussion of the nonlinearities; discuss the tuning procedures used to adjust the electronic-controller gains and the frequency-sweep techniques used to test the valve; present the frequency-sweep data for various excitation amplitudes; and compare the experimental data with results obtained from a computer model.

BACKGROUND

Fuel-Flow Valve Servosystem

The valve was designed to control the main combustor fuel flow in the MGB engine model. The valve needed to have a response rate of 200 Hz when varying the fuel supply at an amplitude of $\pm$ 10 percent of maximum flow. In the test described herein, the valve and its servoactuator were mounted on a bed plate on the floor of a test cell. The purpose of the test was to adjust the controller gains to obtain the highest response from the valve and to verify that at these controller settings the valve would perform as desired.

The basic elements (fig. 1) of the test bed were the fuel-flow valve, the electrohydraulic servosystem, the hydraulic unit, and data-recording devices. The electronic controller for the

fuel-flow servosystem is an analog device for proportional and derivative compensation of the error signal; the larger the error signal, the harder the servosystem is driven as it attempts to follow the desired set point. The compensator output feeds to a power amplifier, which drives a two-stage electrohydraulic servovalve. A linear variable differential transformer (LVDT) measures the actuator-piston position and compares it to the command signal to gauge the error. For safety reasons, air–rather than hydrogen–was run through the fuel-flow valve during the test.

Servosystem Model

The fuel-flow valve servosystem model (fig. 2) was developed to verify the system response obtained during testing. Nonlinear performance limits were accounted for in the design of the model to accurately predict the system response. The physical parameters used to model the servosystem are defined in the table.

The first block in figure 2 represents the proportional and derivative electronic compensator. An open-loop frequency response was performed on the compensator; this response yielded a transfer function of the following form that relates the compensator output voltage $c(s)$ to the error signal input voltage $e(s)$:

$$\frac{c(s)}{e(s)} = K_c(1 + \tau_d s) \tag{1}$$

In the foregoing equation K_c is the compensator gain and τ_d is the compensator-derived time constant.

The output-voltage signal of the electronic controller was applied to the coils of the torque motor mounted on the servovalve. The torque motor coils were modeled as a first-order lag formed by the resistive-inductive impedance of the coils relating $i_c(s)$, the current through the coils, to $c(s)$, the applied voltage as follows:

$$\frac{i_c(s)}{c(s)} = \frac{1}{R_c + L_c s} \tag{2}$$

where R_c is the coil resistance and L_c is the coil inductance. The torque motor coils were rated for 40 mA. Beyond this value, current limiting occurs as noted by the nonlinearity block in figure 2. The effects of nonlinearities on the overall system response are discussed later.

The design of the servovalve model was based on specifications provided by the manufacturer (ref. 1). A second-order linear transfer function relating the hydraulic fluid flow q out of the servovalve to the coil current i_c is shown in the following equation:

$$\frac{q(s)}{i_c(s)} = \frac{K_{sv}}{(1/\omega_{sv})^2 s^2 + 2\zeta(1/\omega_{sv})s + 1} \tag{3}$$

where K_{sv} is the servovalve gain, ω_{sv} is the servovalve natural frequency, and ζ is the servovalve damping ratio.

Hydraulic fluid flowing out of the servovalve was directed to the actuator-piston chamber as shown in figure 1. Actuator-piston resonance can result at high frequencies because of fluid compressibility. The natural frequency of the actuator piston is dependent on the fluid bulk modulus β, actuator-piston area A_a, actuator-piston chamber volume V_a, and load mass M, as follows:

$$\omega_v = \left(\frac{4\beta A_a^2}{V_a M}\right)^{1/2} \tag{4}$$

Using the values in the table and equation (4), the natural frequency calculates to 2500 Hz—well beyond the frequency range of operation during the test. However, a small amount of entrapped air in the hydraulic fluid can significantly reduce the bulk modulus, which in turn will reduce the natural frequency of the valve. In the model, the effects of fluid compressibility within the actuator-piston chamber are considered negligible, and the valve is modeled as a lag that relates the valve position to fluid flow:

$$\frac{x_p(s)}{q(s)} = \frac{K_v}{s} \tag{5}$$

In equation (5), K_v is the fuel-flow valve gain and x_p is the valve position. An LVDT was used to measure the valve position and to convert it to a representative voltage. The LVDT was modeled as a constant gain K_{LVDT}, which relates the valve position in inches to a corresponding voltage.

Nonlinear Performance Limits

Theoretically, the model described should accurately predict the operation of the servosystem. However, the relatively high frequencies at which the test was conducted led to saturation of the servovalve-coil current, and thus, the servosystem was limited in its dynamic response. The effect that saturating nonlinearities have on the servosystem's dynamic response can be predicted by using a technique, presented by Webb and Blech (ref. 2), which consists of deriving two limit lines on the log-amplitude-versus-log-frequency plot that define system operation in the nonlinear region. The first line, the linear-nonlinear transition boundary, separates the linear operating region and the saturation region. The second line, the maximum-performance limit line, defines the operating region beyond which system operation is theoretically impossible. Assuming that coil-current saturation occurs at 40 mA, we obtain the following equations for the linear-nonlinear transition boundary and maximum-performance limit line, respectively:

$$x_p(s) = \frac{0.04 K_{sv} K_v}{s\left[(1/\omega_{sv})^2 s^2 + 2\zeta(1/\omega_{sv})s + 1\right]} \tag{6}$$

$$x_{p\,max}(s) = \frac{4}{\pi}\left(\frac{0.04K_{sv}K_v}{s\left[\left(\frac{1}{\omega_{sv}}\right)2s^2 + 2\,\zeta\left(\frac{1}{\omega_{sv}}\right)s + 1\right]}\right) \tag{7}$$

In equations (6) and (7), $x_p(s)$ represents the valve position at which these limit lines encountered.

TEST PROCEDURE

The first step in the test procedure consisted of adjusting the compensator gains to obtain the best possible valve performance. The controller was set up to sum two external set points. These two inputs consisted of a dc offset and a square wave. Summing these two signals resulted in excitation of the valve around a nominal set point. During tuning, the valve-position feedback was viewed with an oscilloscope to monitor the effect of compensator-gain adjustment on the servosystem step response. Initially, the valve was subjected solely to proportional control. Because of the poor response obtained under this setup, however, the electronic controller was modified to allow the compensator to contribute both proportional and derivative control. This modification resulted in a faster rise time without an increase in the amount of overshoot or oscillation in the system step response—a state of affairs that, in turn, allowed the valve to be driven to higher frequencies with adequate frequency response.

After the tuning of the servosystem, a Hewlett Packard signal analyzer was employed to perform a frequency response test. A dc offset was specified and used to move the valve to the 50-percent-open position, about which the sinusoid could then be oscillated. The source ramp time was specified to control the speed at which source output voltage could change at the beginning and end of a run, thus protecting the valve against sudden position changes. A swept sinusoidal signal was generated to stroke the valve at amplitudes of ±2, ±5, ±10, ±15, and ±20 percent about the 50-percent-open position. The valve position feedback was monitored and compared to the set point to generate the frequency response.

As a result of an excess of stroking cycles, the packing in the valve stem guide wore out during the test. The worn packing allowed the valve stem to move about from side to side within the stem guide, causing excessive wear and galling of the valve stem and guide. In consequence, the packing was replaced, the valve stem and guide were recoated and polished to remove imperfections, and a valve-stem packing retainer was added to secure the packing and reduce side-to-side valve-stem vibration. Thereafter, to guard against damaging the packing, the valve was subjected to a limited number of stroke cycles.

RESULTS AND DISCUSSION

Considerable time was spent tuning the compensator gains to obtain optimal valve performance. As previously noted, the compensator initially was set to contribute only proportional control of the valve; however, it was subsequently determined that the servosystem was not able to provide the desired response under this configuration. Therefore, the compensator was modified to contribute proportional and derivative control. After this modification, a noticeable improvement in the system rise time occurred without an increase in overshoot or oscillation. Figure 3, which shows the system response to a 5-percent step input with proportional control only and with proportional and derivative control, illustrates the improvements.

The results of the open-loop frequency response performed on the compensator after completion of the initial tuning appear in figure 4. As the frequency increases, a noticeable increase occurs in the compensator's control effort. Such an increase is to be expected, since derivative control has a greater effect at higher frequencies. The experimental response was used to develop the compensator transfer function as given in equation (1).

Frequency response data were collected by applying a sinusoidal command signal of various amplitude levels to the controller and sweeping the frequency from 10 to 200 Hz. Figure 5 shows the system frequency response for the five amplitudes at which the test was performed. The amplitudes were normalized with respect to percentage of stroke to which the valve was subjected, allowing the effects of nonlinear saturations to be visualized more readily. At higher excitation amplitudes, limited valve response is marked, as the ± 10, ± 15, and ± 20 percent amplitude responses run up against a limit. Equation 7 was used to develop the torque motor-coil current saturation limit line depicted in figure 5(a). The recorded data show that the servosystem response fell short of the predicted limit line and also exhibited a dip across all five tested excitation amplitudes at 120 Hz, indicating unmodeled servosystem dynamics. Possibly, the nonuniform response was caused by test bed vibration or by entrapped air in the hydraulic fluid. Pressure deviations in the hydraulic supply lines present another possible cause for the nonuniform response. The accumulators mounted on the hydraulic lines were intended to maintain a constant hydraulic-fluid supply pressure to the servovalve; however, the test-bed setup did not permit location of the accumulators as close to the servovalve as desired, thus making hydraulic-line pressure deviations possible. Such deviations would limit the flow into and out of the servovalve and limit the actuator-piston position.

Figure 5(b) depicts the phase response for the five tested excitation amplitudes. The phase responses also show the marked effects of nonlinearities at the higher excitation amplitudes. A design requirement of the valve was that it exhibit a 200-Hz response rate in the presence of a fuel supply varied at an amplitude of ± 10 percent of maximum flow. During testing, the servosystem exhibited a dynamic response of 0.708 of the commanded amplitude (-3 dB) at 190 Hz for a commanded amplitude of ± 10 percent of full stroke—acceptable for the MGB engine model test. Frequency-response tests were performed with and without air passing through the valve, and, as expected, no difference in frequency response was recorded.

In figure 6 of the model's predicted response is compared to the experimental data to demonstrate the model's validity. Figure 6 shows the plot of the normalized fuel-flow valve position for an excitation amplitude of ± 10 percent over the frequency range of the test as predicted by the model and found in the experimental data. The model predictions cover the effects of coil-current saturation limiting. The plot shows fair agreement between the model and the experimental data, with the discrepancies between the two attributable to previously mentioned unmodeled servosystem dynamics, such as test-bed vibration or pressure deviations in the hydraulic supply lines. Similar agreement is exhibited between the experimental data and the model for the remaining four excitation amplitudes.

CONCLUDING REMARKS

The analysis and performance of an electrohydraulic servosystem for providing high-frequency positioning of a fuel-flow valve has been presented. The servosystem was described, and a model was developed that included the effects of saturating nonlinearities. A test bed was set up to tune the electron controller compensator gains and to analyze the system's response to swept-frequency inputs. The compensator initially was set to contribute only proportional control; modifying the compensator to contribute both proportional and derivative control resulted in a better response. The recorded experimental data showed that the effects of saturating nonlinearities became noticeable when the servosystem was driven out to higher frequencies; this was found to be particularly true for large excitation amplitudes. The validity

of the developed model was demonstrated by showing that it had reasonable agreement with the recorded experimental response.

REFERENCES

1. Moog 760 Two-Stage Flow Control Servovalves. Catalog 762 682, Moog Inc., Industrial Division.

2. Webb, J.A., Jr.; and Blech, R.A.: Predicting Dynamic Performance Limits for Servosystems with Saturating Nonlinearities. NASA TP-1488, 1979.

TABLE 1. - FUEL-FLOW VALVE SERVOSYSTEM PARAMETERS

Controller gain, K_C, V/V	1.144
Controller derivative time constant, τ_d, sec	4.082×10^{-3}
Servoamplifier coil resistance, R_C, Ω	40
Servoaamplifier coil inductance, L_C, H	0.18
Servovalve gain, K_{SV}, in.^{3}/sec·A	964.0
Servovalve natural frequency, ω_{sv}, rad/sec	$2\pi(160)$
Servovalve damping ratio, ζ	1.41
Fuel-flow valve gain, K_v, 1/in.^{2}	1.812
Load actuator piston area, A_a, in.^{2}	0.552
Total hydraulic fluid volume in actuator, V_a, in.^{3}	0.276
Load mass, M, lb·sec^2/in.	3.571×10^{-3}
Hydraulic fluid bulk modulus, β, lb/in.^{2}	2.000×10^5
Hydraulic fluid density, ρ, lb·sec^2/in.^{4}	9.350×10^{-5}
Hydraulic supply pressure, P_{SU}, lb/in.^{2}	3000
LVDT feedback gain, K_{LVDT}, V/in.	18.6

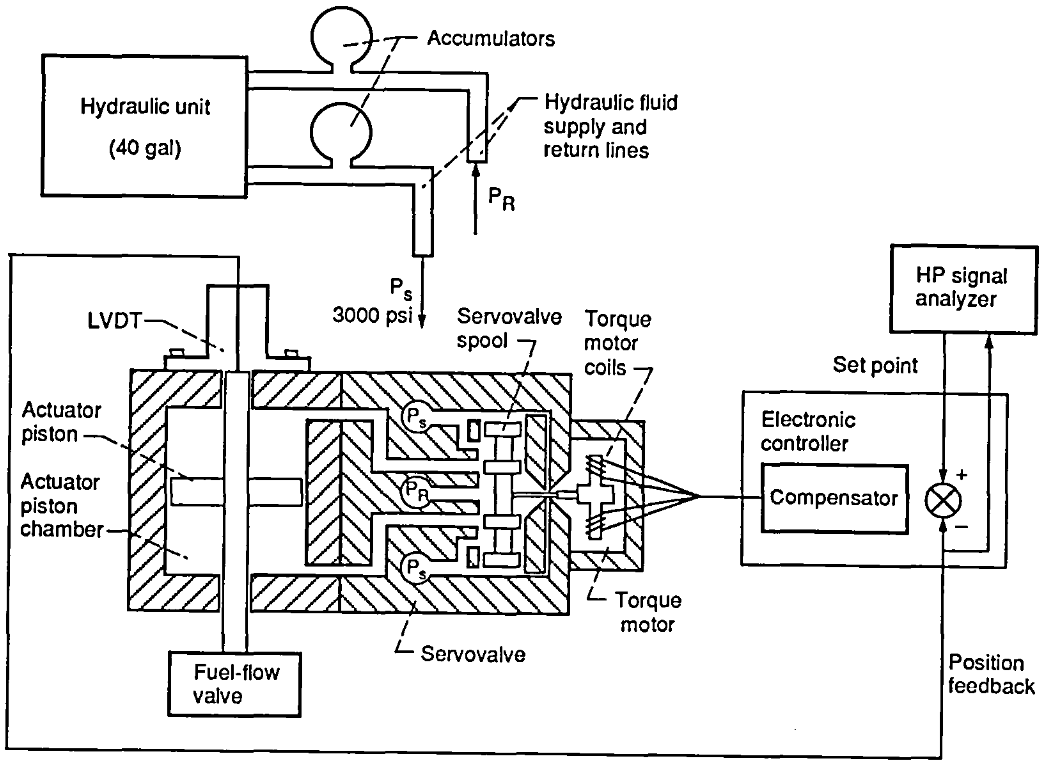


Figure 1.—Fuel-flow valve test bed.

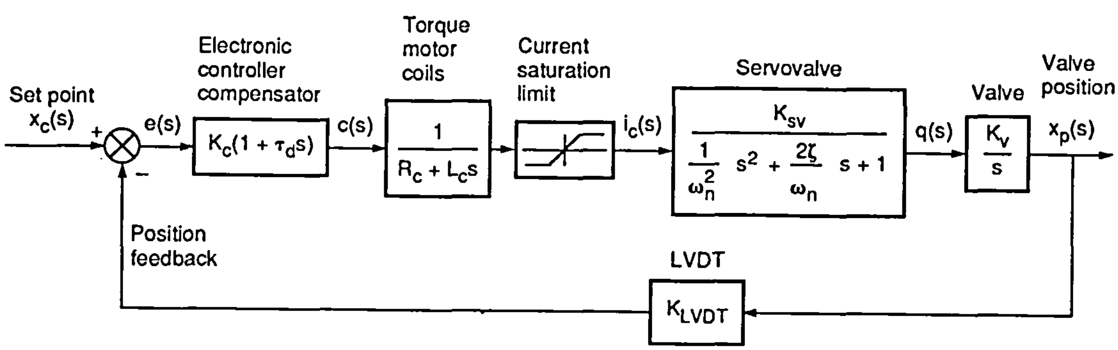
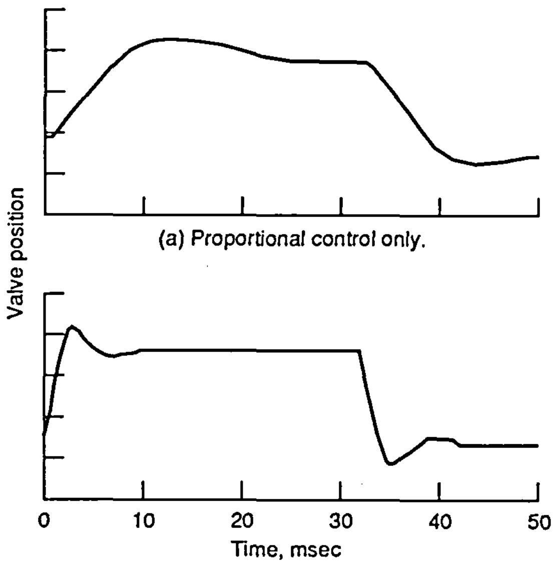


Figure 2.—Servosystem model.



(b) Proportional and derivative control.

Figure 3.—System response to 5 percent of full-stroke step.

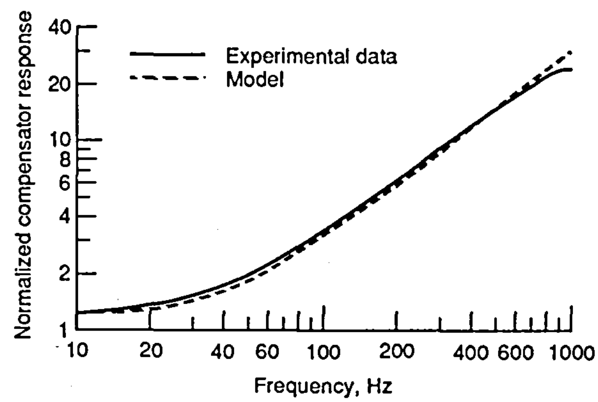
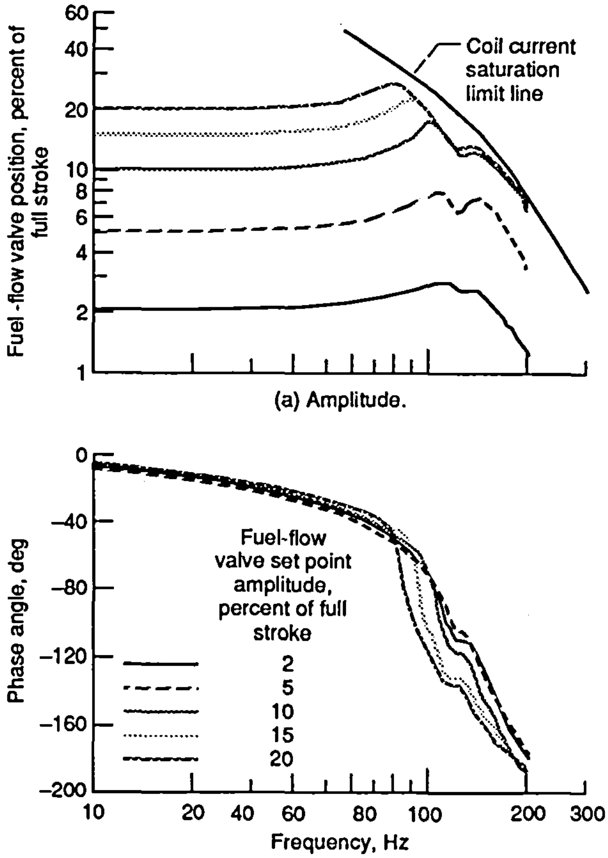


Figure 4.—Compensator open-loop frequency response.



(b) Phase.

Figure 5.—Servosystem experimental closed-loop frequency response, $x_p(s)/x_c(s)$.

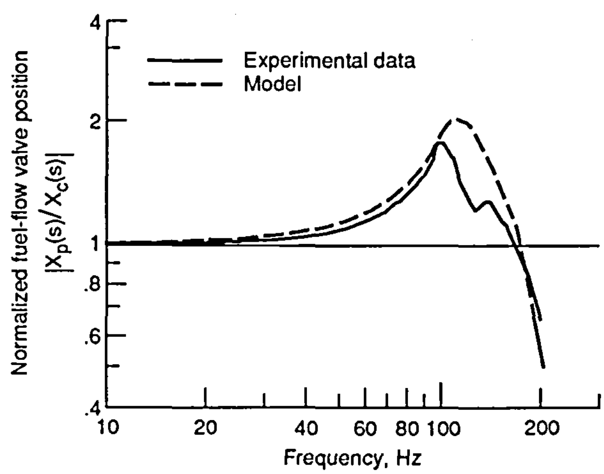


Figure 6.—Comparison of servosystem response with model at 10 percent of full-stroke amplitude.

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